

A Split-Single Two-Stroke Engine

An efficient unit for propelling a class "C" Hydroplane

by R. E. Mitchell

THE American design and British copies of what has now become known as the "modern racing engine" suffers from one major defect. It is unreliable. No doubt the high output figures at speeds of the order of 15,000 r.p.m. are realised on the test bench where the load may not be applied until these high speeds are reached and the jet needle can be very carefully adjusted in order to give the maximum output.

When one of these engines is put into a car or a boat, the majority cannot be persuaded to give of their best. The rules governing model car racing allow as many laps start as the owner desires to attain maximum speed, or if the engine is particularly slow off the mark the car may be lead manually by means of the tethering cable. In boating circles conditions are considerably more difficult. Half a lap (i.e. 50 yards) is all

that the rule allows before timing begins. Unlike cars, the highest demand on the engine is at the start where the propeller blades are working at an unfavourable angle of attack and the hydroplane is behaving as a displacement boat. To avoid this the technique appears to let the engine race at very high speed, which can only do it considerable harm if not wreck it altogether and hurl the boat forward hoping that the engine will maintain its high revolutions. More often than not the launch is faulty or the surface of the water is blamed for the fact that the revs. do drop and the engine splutters its way round the course or fails to complete the distance. Starting this type of engine also appears to be extremely difficult and is possibly the reason why mechanical starters have become popular.

If the published b.h.p. curves of the engines in question are examined they only start at about 8,000 r.p.m. and when they are produced to meet zero they give a complete curve of the type shown in Fig. 1. Compared with the curves published for lapped piston engines, also shown in Fig. 1,

they show that at the lower speeds the engines fitted with ringed light alloy pistons show a surprising lack of power. It may be argued that full-size racing engines show a similar tendency but here drivers and gearboxes can ensure that the r.p.m. are kept high. These engines also start with comparative ease. It is possible that the poor startability and low speed characteristics

are due to the aluminium alloy piston's inability to seal effectively the transfer port from the exhaust port, at least when the engine is relatively cool. The ports have crept further and further round the cylinder periphery to increase the power at the top end of the scale, losing sight of the fact that the engine has to attain these revs. to be of any use. The lack of power is not apparent in engines fitted with solid steel or cast-iron pistons because here the piston clearance

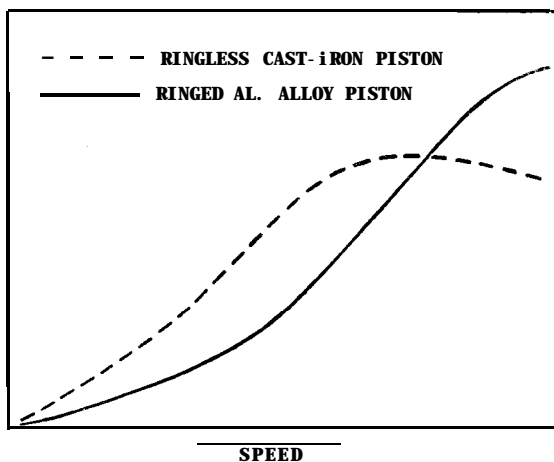


Fig. 1. Power outputs of plain cast-iron piston, and ringed light-alloy piston

is of the order of one-tenth of that which has to be given to aluminium alloy pistons to allow for expansion at the working temperature. The clearance allowed for the rotary inlet valve is not such a bad offender because this is more effectively sealed by the liquid fuel/oil mixture. A fact which supports this theory is that an engine of this type is easier to start with glow-plug than if spark-ignition is employed. With spark-ignition, by the time there is enough fuel and oil to effectively seal the ports the plug electrodes have become drowned so that no spark takes place. At M.P.B.A. regattas silencers are compulsory and these are objected to by some users of this type of engine because it is more difficult to get rid of the excess fuel after flooding so that the process may be repeated. Also, the familiar expedient of priming through the exhaust port cannot be done so easily in these circumstances.

The starting difficulty is not due to an over-rich mixture since with methanol (the most usual fuel) the correct mixture is obtained at 65 deg. F. while the lowest combustible mixture occurs at

45 deg. F. A glow plug, on the other hand, will continue to glow even in such wet circumstances and the engine will consequently start easier. Starting the car on its run appears to be done on an over-rich mixture and it is allowed to run until the mixture has weakened by which time the speed may have been built up. It is possible that the high centrifugal forces, which are set up on small circular tracks, acting on the fuel level, thus giving a large variation in mixture

of the scale have been improved. The engine referred to is that fitted to Sparky II and built by Mr. G. Lines, of Orpington.

To readers of Mr. Westbury and regatta reports in THE MODEL ENGINEER since the Class "C" (10c.c.) hydroplane was introduced, it will be noted that this type of craft is usually extremely unstable. It has been argued that the high power/weight ratio is responsible. Under the usual regatta conditions obtaining in this

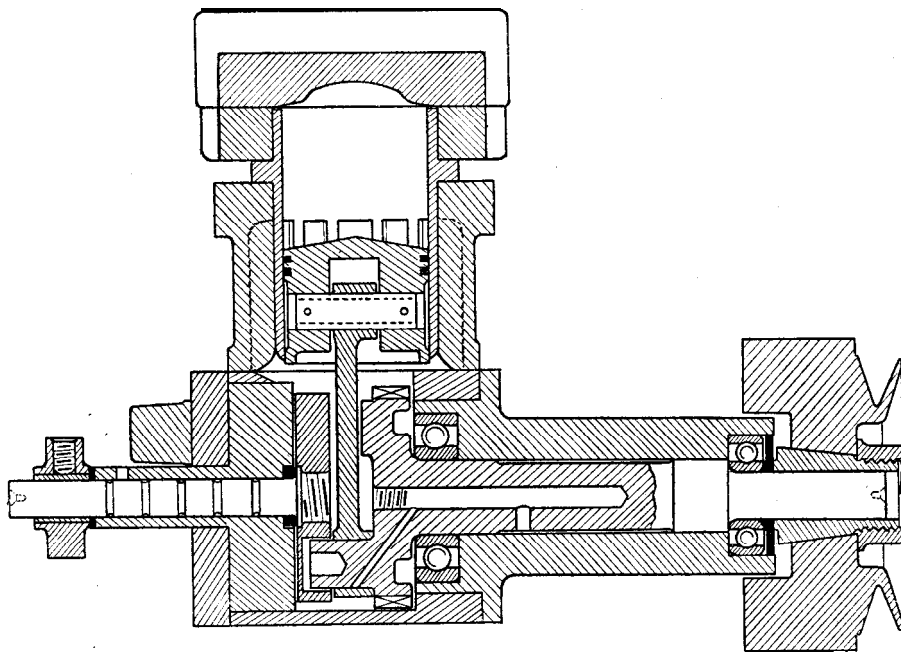


Fig. 2. Longitudinal section through transfer cylinder and crankshaft; full size

strength, may be a decided advantage. Somewhere during the run the mixture is slightly on the rich side to give the maximum power output so enabling the quarter mile or so to be timed. In the hydroplane field the centrifugal force is considerably lower and its effect is not so pronounced, with the result that the whole run is most likely done with a very rich mixture and maximum power is never attained. A boat will occasionally run out of fuel a lap or so after the timed distance and, in some instances, the speed will increase considerably for the last fifty yards or so immediately before the fuel tank is empty.

It is quite likely that such a boat would have petered but had the mixture been weakened at the start. The overall performance of one well-known hydroplane engine has been improved out of all recognition by replacing the aluminium alloy piston with rings by a piston skirt of cast-iron minus the rings. It is more than likely that the maximum output has been reduced by an increase in the mechanical friction, but very definitely output and reliability at the lower end

of the scale this is not the case. It is not certain what the effect of the weight of the boat is on the power required to drive it.

The Class "B" hydroplanes, *Beta* and *Beta II*, were powered by the same o.h.v. four-stroke engine which, when first assembled, had a power output of slightly over 1.5 h.p. It is quite possible that with subsequent running this has increased somewhat. Both these boats were fairly heavy for their class, being of the order of 8-1/4 lb. and were propelled fairly consistently in almost any conditions at about 45 m.p.h. The average Class "C" hydroplane weighs no more than half this figure, but speeds at regattas above 45 m.p.h. are a rarity. Therefore their engines could not have been developing their full output of 1-1/4 h.p. + of which they are claimed to be capable. There is such a wide variety of hull design that the hull cannot be held responsible. Under competitive conditions the roughness of the water is the same for all classes of boats and obviously the heavier the boat the more stable it can be expected to behave. The introduction of surface propellers has also increased

the instability of hulls, but this has been more pronounced in Class "C" because since the boat is inherently more unstable the propeller leaves the water more frequently. This means that the torque reaction on the hull and the side thrust on the stern, obtained by the use of surface propellers, are alternatively applied and removed as the hull bounces in a vertical plane. Also, when the propeller is thrown clear of the water the thrust is removed which further increases the

and one combustion chamber are used. The intention, originally, was to use two carburettors ; but this was not considered very practical, so instead, a manifold connecting the two inlet ports is used. It varies from the more usual design of this type of engine in that there are two crankshafts which are synchronised by making the crank discs a pair of spur wheels. Internal gearing is preferable since general engine "petrol" lubrication can take care of them.

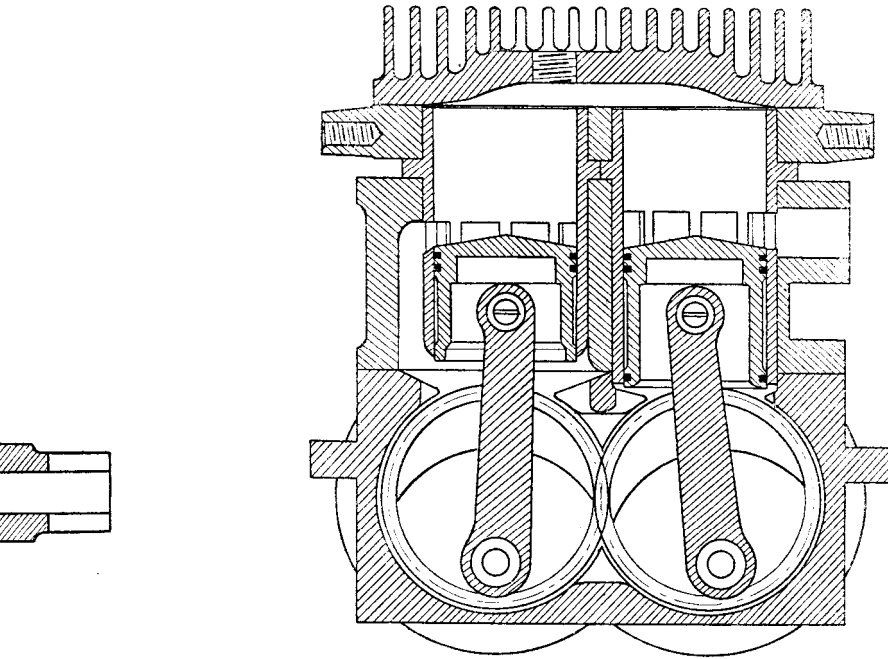


Fig. 3. Transverse section through both cylinders ; full size

vertical instability. This type of instability can be minimised by the careful attention to the planing angles (most appear to fit planes at too steep an angle) and the position of the centre of gravity. It follows that a large contribution to stability could be made by the removal of side thrust on the stern and the torque reaction on the hull. The only method of eliminating those effects would be to fit two propellers running in opposite directions. Any method of transmitting the power to two shafts through gears from one engine was considered to be too inefficient. Using two engines with synchronising gears on the flywheels was considered. Since each engine would be of 5 c.c. capacity and almost certainly a two-stroke in this size it would probably suffer from the faults previously assigned to this type of engine. A convenient way out was found in which essentially two engines are used, and is a variation of a principle used in several full-size designs. The general arrangement of the engine is shown in Figs. 2 and 3.

It consists of duplicating everything except that only one transfer passage, one exhaust port

The slight displacement of the pistons, in which one controls the transfer port while the other takes care of the exhaust can be conveniently achieved by orienting the gear teeth differently on the crank discs. Choosing a suitable number of teeth governed by the angle of displacement required and displacing the wheels by one tooth will give a similar effect. If the constructor makes his own gearcutting tools a non-standard diametral pitch can easily be used, and this was the method adopted in the present case. A usual timing of port events for a two-stroke may be to open the exhaust port 65 deg. before bottom dead centre while the transfer opens 55 deg. before b.d.c.

The closing takes place at similar angular displacements after b.d.c. This type of timing is shown in Fig. 8. With the simple two-stroke it is impossible to vary the symmetry of the timing which is not the most efficient that could be desired ; the transfer possibly opening too soon after the exhaust while some charge is possibly lost by closing the exhaust port after the transfer port has closed. Considerable experi-

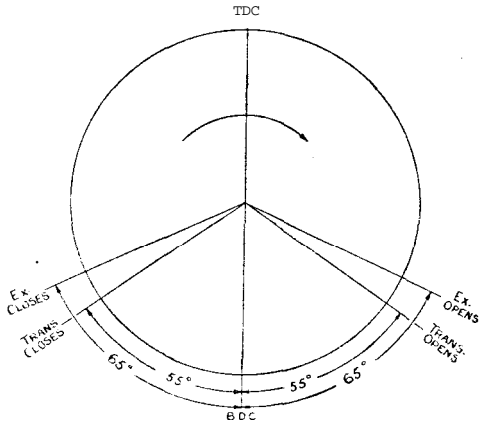


Fig. 8. Orthodox two-stroke timing

ment with deflector and cylinder-head design appears to be necessary if this is to be prevented. In the two piston designs where one piston is responsible for the transfer while the other piston takes care of the exhaust, it is an easy matter to upset the symmetry by advancing the exhaust controlling piston by a suitable amount. Using the timing given above, it was considered that a crankshaft displacement angle of 10 deg. would be adequate. This enables the exhaust port to be opened 20 deg. before the transfer port opens while they both close together. This displacement means that both pistons are not at top centre together but from previous experience a difference of ± 5 deg. in the ignition timing can be regarded as insignificant. Fig. 9 shows

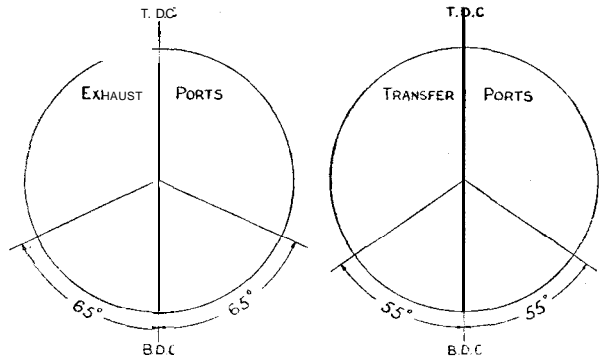


Fig. 9. Individual transfer and exhaust timing used for split-single

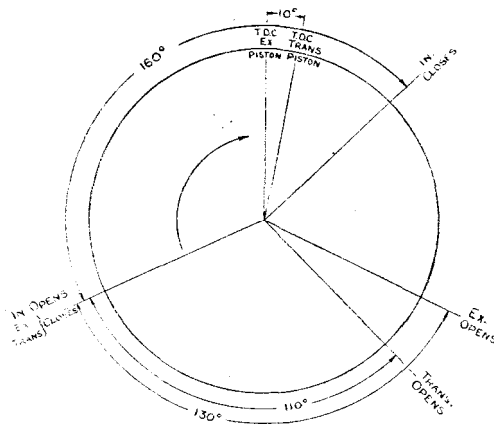


Fig. 10. Combined timings show in Fig. 9 used for split-single

the individual timing of the transfer and exhaust ports, while a combination of the two is shown in Fig. 10. The out-of-phase angle of 10deg. means that 36 teeth are required on the crank discs. This will enable the conventional symmetrical timing to be used, if necessary, to compare with the intended timing.

One difficulty which presented itself was the making of the cylinder-head joint. This would be easy if the cylinders were in one block, but with monoblock construction the provision of adequate transfer and exhaust passages would be very difficult. These could be secured to the outside of the cylinders by, say, brazing, and although ports brazed to cast-iron have been used by various constructors it is not a sound method, particularly where vibration is present. Clamped-on ports which have previously been used in two-stroke design were dismissed as not being satisfactory to use in the present case. If cylinder liners are used, which makes the cutting of transfer and exhaust passages in the cylinder block an easy operation, the joint cannot

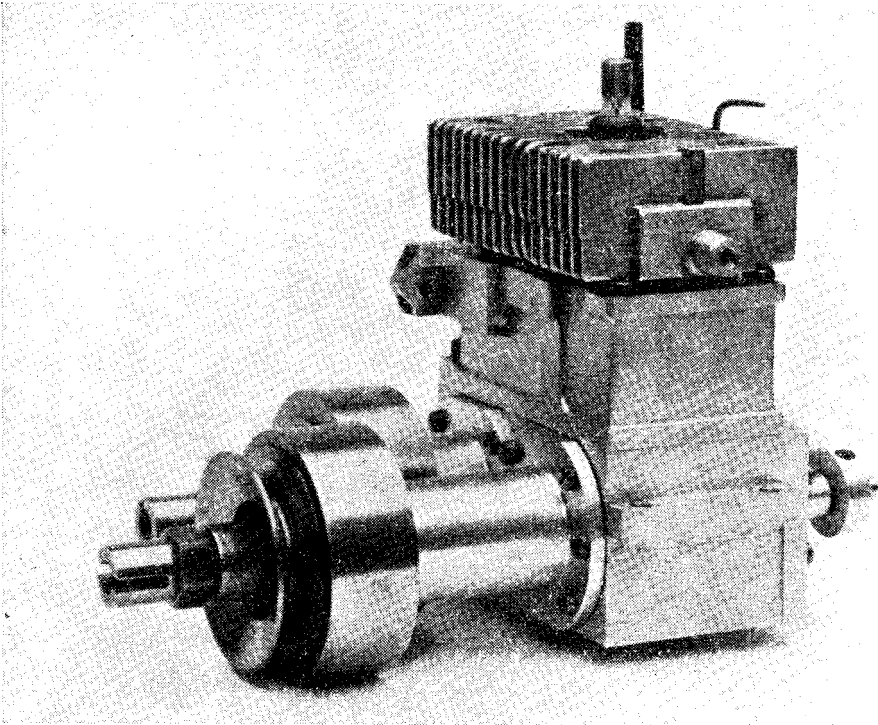
be made on the top end face of the liners due to the provision of a common combustion chamber being imperative.

The difficulty was overcome by making a port belt for the two cylinders with ports coinciding with those in the liners. The liners each has a narrow flange which rests on the top of the port belt. Over the protruding portion of the liners is placed a second belt which is provided with cooling fins, the thickness of which is slightly greater than the upper portion of the cylinder liners. On top of this the cylinder-head made its joint. It will be seen that the joint on the liners is made at the flanges between the top of the port belt and the bottom of the finned belt. This necessitated having flanges of exactly the same depth but since the actual dimension is not critical this was easily carried out as described later.

A further advantage of this design is that a deflector on the piston is not required since the incoming gas is automatically directed up the transfer cylinder into the combustion chamber

and down into the exhaust cylinder. In order to help the charge in the correct direction both the transfer and exhaust piston crowns are made conical, the angle of which is very obtuse. Further, there is no possibility of leakage from the transfer to the exhaust port around the piston

The rotary valves call for no comment except that they consist of mild-steel discs on nickel steel spindles which run in dural bearings. One of the rotary valve spindles extends beyond its bearing in order to take a coupling for a magneto drive. The usual 180 deg. induction period



An external view of the finished engine

skirt and the ports occupy the whole of the periphery of the liners except for a small portion where they are adjacent, there being no space available at this spot and it is an advantage to have the cylinder centres as close together as possible. The actual portion of the circumference used for ports is go per cent. Electron pistons are used, as in all previous designs to keep the reciprocating weight to a minimum. Two compression rings are fitted to each piston. Since these pistons have to be fitted with large expansion clearances the exhaust piston is fitted with a third ring at the bottom of its skirt to prevent leakage of the charge from the crankcase to the exhaust port.

It was rather a problem to decide on a suitable bore and stroke as obviously the centre distances of the cylinders determine the bore while the stroke is governed by crankshaft centres. Actually, to avoid too large a bore/stroke ratio, the cylinders were made desaxe to the extent of 1/16 in. which E. T. Westbury has often recommended. The final dimensions arrived at were 0.740 in. bore by 0.700 in. stroke which gives a total displacement of 9.8 c.c.

was considered to be excessive in view of the fact that the transfer port closes rather later than the single-cylinder engines and thus was reduced to 160 deg. The opening of both valves takes place at the same time at which the transfer and exhaust valves close.

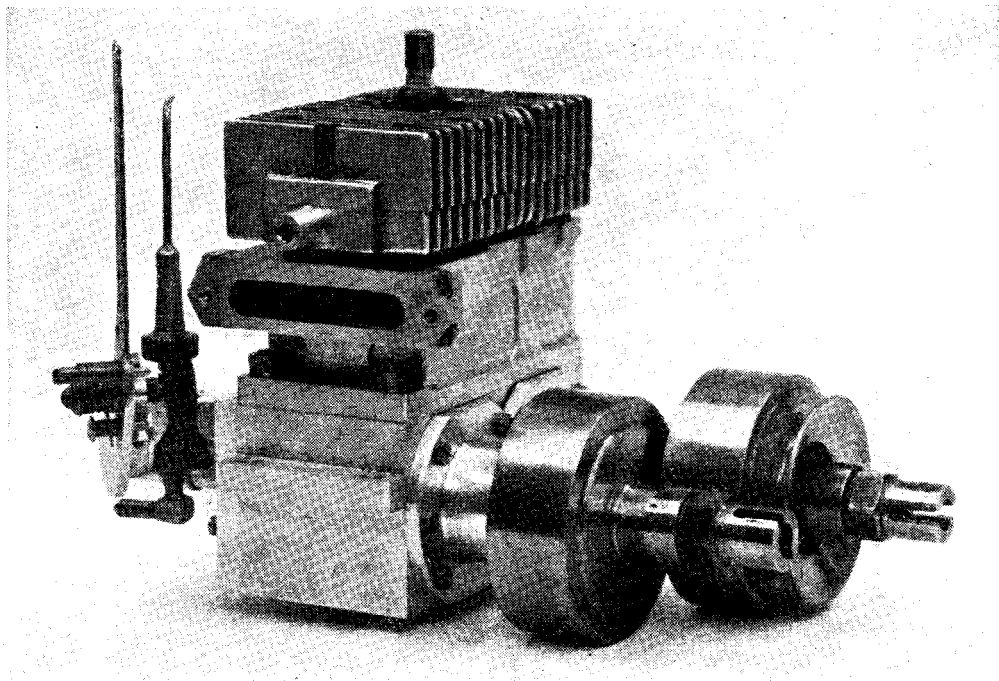
For a twin-screw boat the propeller shafts would not be parallel and an attempt was made to design the engine so that the crankshafts would be at the same angle as the propeller shafts, thereby avoiding the use of universal joints at the inboard end. This would entail the making of obtuse angle bevels for the synchronising gears and complicated angular bolting faces would also result. Also the rotary valves would be reduced in diameter. The straightforward method was adopted and two universal joints per shaft are used. In order to obtain a useful flywheel diameter it is necessary to stagger them one behind the other and provide crankshafts of different lengths to accommodate them. The flywheels, which are of mild-steel, are secured to the shafts by means of taper split collets, the female portion of the universal joint acting as a nut.

(To be continued)

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IT will be noted that the plug is offset towards the transfer cylinder. This was done, since in this position the charge is in all probability less diluted with exhaust gases and consequently will be easier to ignite. In order to prevent gas escaping into the ports before the correct opening time due to large clearances between the cylinder wall and piston, the top ring is placed as near to the top of the piston as is thought practical and this same distance is used between the rings. The transfer cylinder piston is machined away for lightening purposes on the inside, since the outside is in contact with the transfer port and crankcase volume is not therefore reduced as in the case of the exhaust piston where the outside of the piston is isolated from the crankcase. In this case the lightening is done on the outside of the piston. Case-hardened hollow gudgeon-pins

are used, but they are not fitted with end pads. Previous experience with end pads has shown that they can cause *severe* scoring of the cylinder bore. Endwise location is secured by means of piano wire pins through holes drilled in both gudgeon-pin and piston. The pins are secured by having their ends bent over at right-angles.

The crankshafts, which are of the usual overhung type with the crankpins sufficiently long to provide a drive for the inlet valves, are machined from 3 per cent. nickel-steel and are case-hardened on completion. Care had to be exercised in the heat treatment to prevent distortion, since no grinding equipment was available. Lapping of the main journals and crankpin to diameter was the only work carried out on the crankshafts after hardening. The shafts were drilled with the idea of supplying oil, by means of a pump, to the big-ends at a later date.

Complete balancing was not found to be pos-

*Continued from page 475, "M.E.," April 10, 1952.

sible owing to the fact that the periphery of the crank discs has to be complete to receive the synchronising gear teeth. Actually, about one-quarter of the reciprocating weight is balanced instead of the usual half. In passing, complete balancing of the rotary valves was not obtained. The three flat-bottomed holes, drilled nearly through the thickness of the disc and covered by thin discs of brass soldered in, hardly proved

been found that flat surfaces provide better bolting faces for additions that are sometimes found necessary in development work of this nature.

The first parts to be tackled were the crankshafts which are shown in Figs. 11 and 12. This was done because it was felt that, with the crude gear-cutting equipment available, the exact centre distance of 1-3/32 in. of the crankshafts

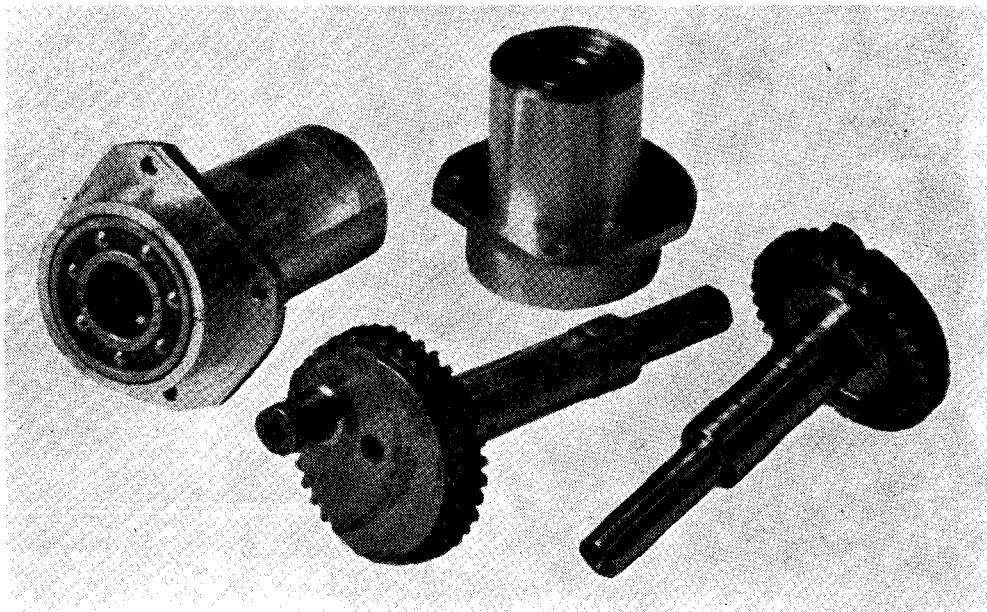


Fig. 11. Crankshafts and main journal housings

sufficient and the peculiar shape of the discs in the vicinity of these holes is the result of filing away metal to obtain as good a balance as possible. Reducing the thickness of the disc opposite to the cut-away portion, at the expense of increased crankcase volume, would, most likely, have been a better method. The out-of-balance forces here would have been less had aluminium alloy discs been used. These were decided against because trouble, in the form of severe scoring, had previously been encountered when attempting to run dural on dural. All the light alloy parts, which include the crankcase, front and rear crankcase covers together with the stationary portions of the inlet valves, the port belt, cooling fins, cylinder-head and carburettor, are machined from the solid bar. The material is an aluminium alloy of the duralumin type to B.S.S. 6L1 obtained from Messrs. Rollett, of Liverpool. This material gives a minimum tensile strength of 24 tons per sq. in. compared with 9 tons per sq. in. for the more usual casting alloy DTD 424. The common fault of porosity with aluminium alloy castings is also avoided and it is doubted whether the making of patterns for one-off jobs is worth while. The angular appearance of the engine could have been avoided but it has

could not be guaranteed. Three per cent. nickel case-hardening steel has been used previously for crankshaft construction with excellent results. The bar was gripped in the chuck by the end to be later machined for the crankpin. It was then reduced to 1.14 in. diameter which is the outside diameter of the gear teeth. The journals of 3/8 in. and 1/4 in. respectively were then turned 0.001 in. oversize to allow for lapping after case-hardening. A 1/16 in. radius is provided at the corners to eliminate, as far as possible, subsequent fatigue failure. Sharp corners act as "stress raisers," and fatigue cracks can easily spread from such abrupt changes in cross-sectional area. For a similar reason all highly stressed parts are given as good a surface finish as possible to eliminate circumferential grooves left by the turning tool. Stub teeth, which have been used for camshaft drive gears in previous four-stroke designs, were decided upon, using 36 teeth. A gear wheel of almost these dimensions was found and a single point high-speed steel tool was ground, using this wheel as a gauge, making slight modifications as were thought necessary.

The tool was mounted on its side in the lathe tool holder and the teeth were planed by means of the rack and indexing from a drilled plate

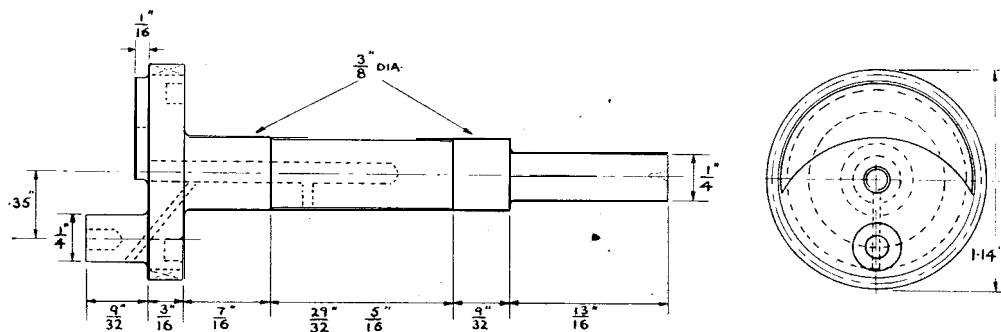


Fig. 12. Details of crankshafts

secured to the end of the lathe mandrel. The teeth were finished to within about 0.002 in. of the correct depth before passing on to the next tooth. The final finishing was done afterwards in one cut without moving the top-slide. The component was then reversed and the 3/8 in. diameter portion was centred by inserting in a hole of the correct size bored through a piece of 1/2 in. diameter brass rod gripped in the three-jaw chuck, and the chuck key was given an extra turn after inserting the crankshaft. Had collets been available this operation would have been easier. The crankshaft was then drilled 7/64 in. diameter to provide an oil way for big-end lubrication. Also at this setting the portion which had previously gripped in the chuck was machined true. The actual diameter here is not impor-

tant provided that it is larger than the engine stroke plus the crankpin diameter. At this stage, two 3/8 in. diameter holes were bored in a block of brass at the correct centre distance intended for the crankshafts. This was carried out in the vertical slide using the cross-slide index to obtain the required centre distance. From this it was found that the gears worked together satisfactorily, although the back lash was rather more than could have been desired.

To machine the crankpins and balance weights the components were mounted in turn on a Keats vee-angle plate secured to the lathe faceplate. Great care has been taken to obtain the correct orientation between the gear teeth and the pin to obtain the designed 10 deg. lead of the exhaust piston. The correct eccentricity to give

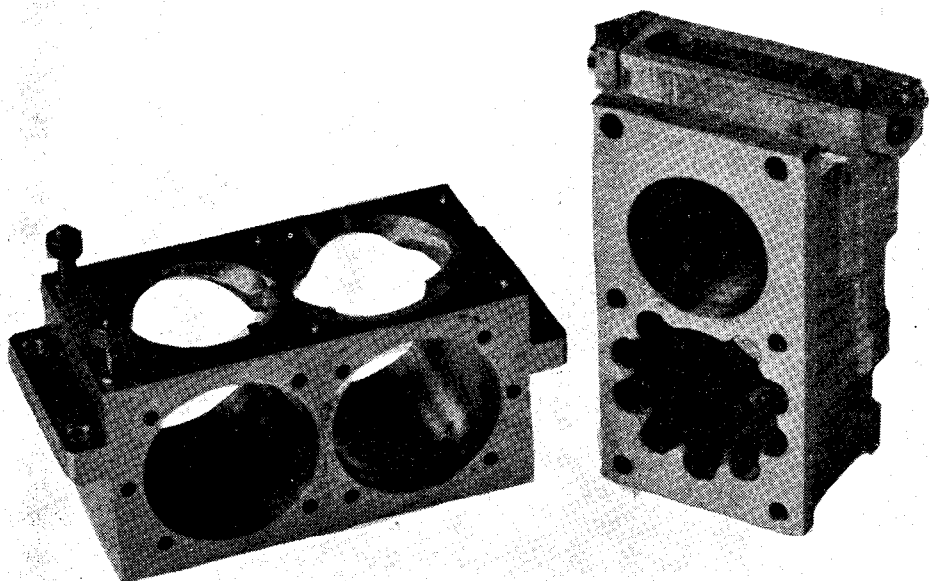


Fig. 13. Crankcase and port belt

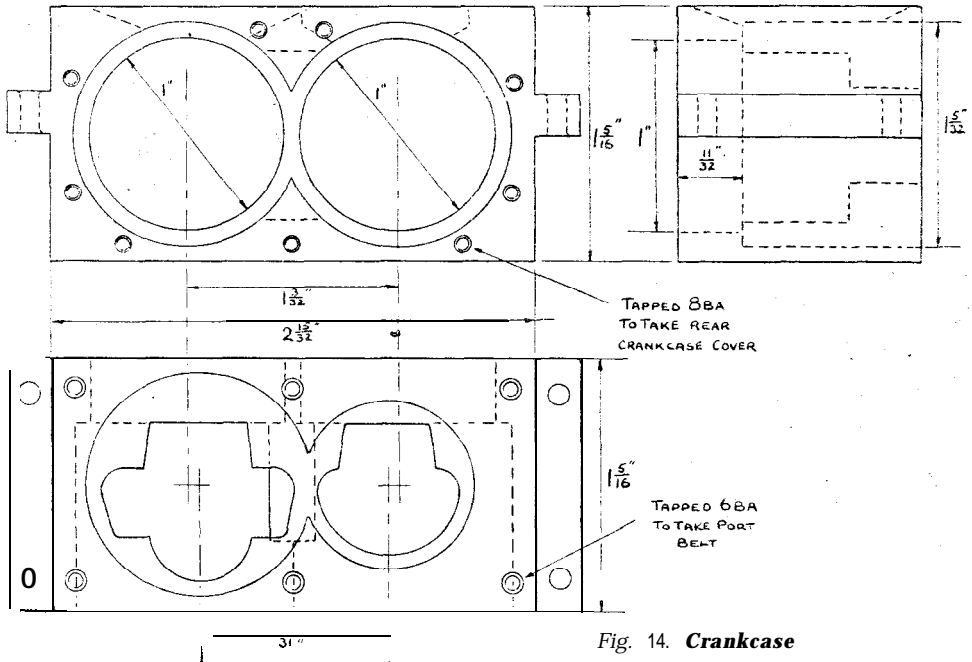


Fig. 14. Crankcase

a stroke of 0.700 in. was measured by means of a dial gauge. The crankpin, which is 1/4 in. diameter, was a straightforward job but owing to the intermittent nature of the cuts, for the most part, these were only light to avoid the risk of the crankshaft turning in the vee of the angle-plate. A 1/16 in. radius was left where the crankpin joins the crank disc and a 7/64 in. diameter drilled up the centre to a depth equal to the length required to drive the rotary valve. A hole of 1/32 in. diameter was drilled diagonally through the crankpin and crank disc to connect with the central 7/64 in. diameter hole. After the machining had been completed the crankshafts were pack carburised at 920 deg. C. to give a total case depth of 0.010 in. Experiments were first carried out to determine the length of time required, because too great a depth results in the whole tooth section being hard and, consequently, brittle. Experience of this had been obtained previously when the pinion driving the camshaft of a four-stroke shed its teeth very early in its career. Hardening was carried out by quenching in oil from 800 deg. C. from an electric muffle. A light temper for half an hour at 150 deg. C. in an oil bath completed the heat treatment. The diameters were then finally lapped to size using a split external lap of aluminium held in a die holder. The journals were made a fairly tight fit in the bores of the ball races.

The next part to be tackled was the crankcase, which is shown in Figs. 13 and 14, having decided that the designed centre distance of the crankshafts has been adhered to. It was roughed out by sawing from 2 in. sq. duralumin bar.

The length was made equal to the width of the crankcase over the holding-down lugs plus about 3/8 in. The block was then machined all over

except for the ends, making the faces as flat and square as possible. Four holes were drilled and tapped 4 B.A., one at each corner of one of the faces. The reason for this was to secure the block to a piece of 2 in. x 1/4 in. ground mild-steel bar which was, in turn, bolted to the faceplate. Four similar holes were drilled in one of the adjacent faces, to enable the block to be turned through a right angle. This enabled the use of an angle-plate to be dispensed with since they are rather clumsy when used on a small lathe. The 3/8 in. extra length of the block accommodates the tapped holes which are cut away when finally machining the ends of the crankcase. The two bores to take the main bearing housings were bored in the lathe with the work on the faceplate, being set over by the correct amount for the second bore. It will be noted that the larger holes bored in the crankcase overlap.

This is necessary to accommodate the synchronising gears and to provide communication between the two portions. Owing to this overlap the crankshafts have to be removed from their housings before these are removed from the crankcase. In the more usual design the housing with crankshaft in situ can be removed from the crankcase. No further facing up of the external surfaces was attempted. Using the other set of tapped holes, the crankcase was turned through 90 deg. and the holes to receive the bottom ends of the cylinder liners were similarly bored. The centre distance in this case being 31/32 in. At this stage the clearances for the connecting-rods were milled. The holding-down lugs were formed on the ends of the crankcase by milling with the work held in the vertical slide.

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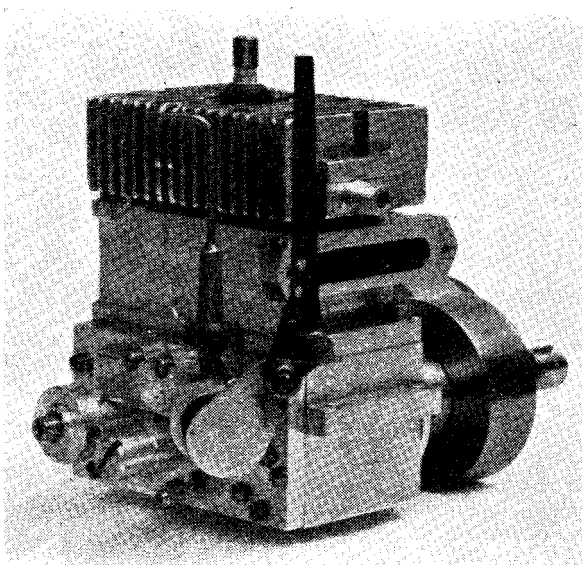
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THE two main journal housings were the next components to be made and are shown in Figs. 11 and 15. These were machined from $1\frac{1}{2}$ in. diameter duralumin bar.

A suitable length was cut off and bored and reamed, longitudinally, to a diameter of $\frac{3}{8}$ in. right through. At this setting the external diameters were also turned together with the bore to receive the smaller ball-race at the flywheel end. The work was then reversed and placed on a $\frac{3}{8}$ in. diameter mandrel prior to machining the register at the crankcase end and



forming the recess to take the larger ball-race also at this end. Behind the ball-race was machined a small recess to aid in its removal should this become necessary.

The flanges were marked off for No. 43 diameter holes to take 8-B.A. screws to secure the housings to the crankcase. The flywheel clearance on the longer component was machined by mounting it eccentrically on the faceplate by means of a through-going bolt. The ball-races were

inserted by heating the housings up to 100° C. or so, at which temperature the races dropped into position to be held tightly on cooling.

Figs. 13 and 16 show the port belt which was started as for the crankcase in that a block of

**Continue d. from page 506, "M.E.," April 17, 1952.*

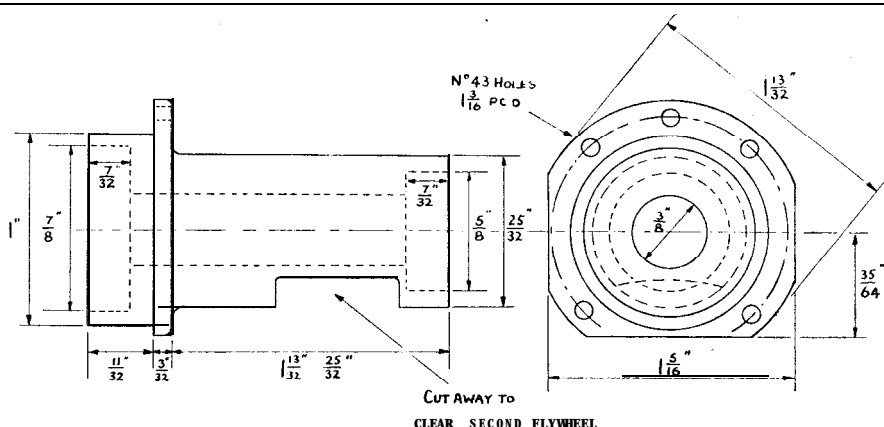


Fig. 15. Main journal housings

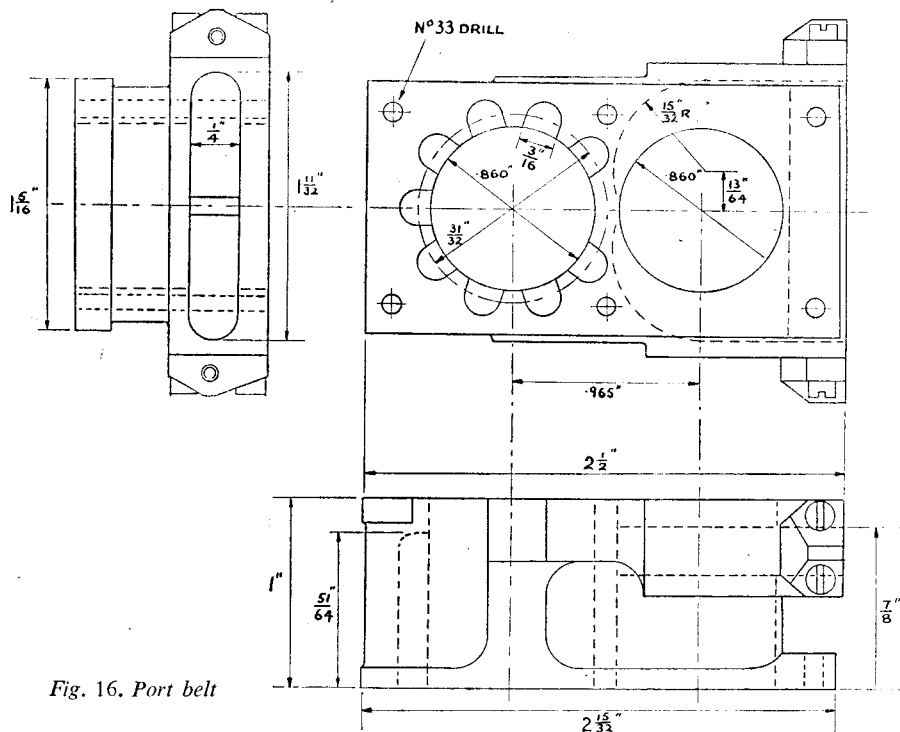


Fig. 16. Port belt

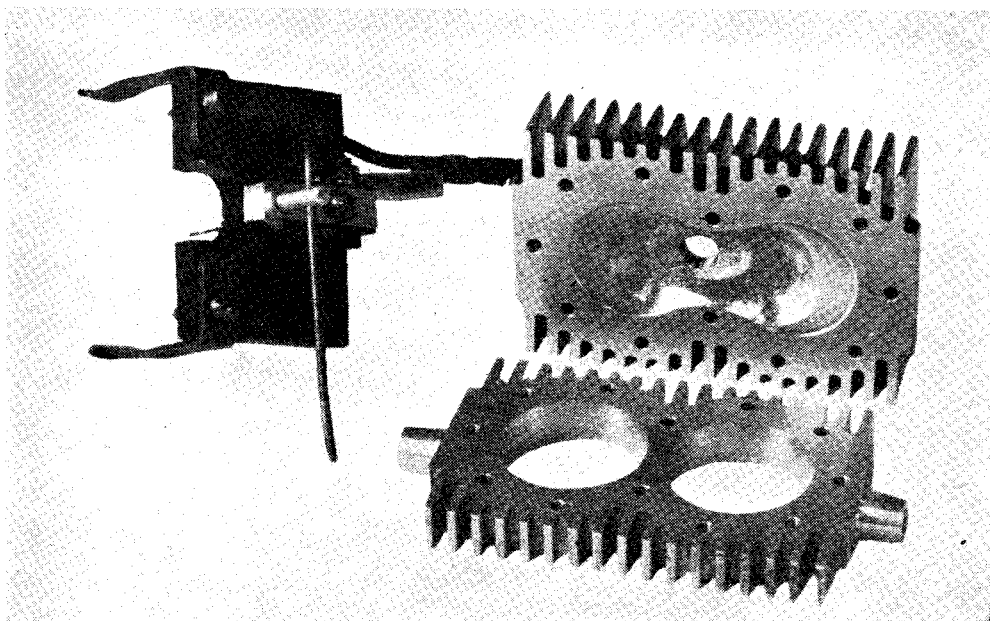


Fig. 17. Cooling fins, cylinder-head and glow-plug connector

duralumin with square flat surfaces was first prepared. One face was marked off for the cylinder liner bores and the transfer passages, which consist of nine $\frac{3}{16}$ in. diameter holes, were drilled on a $\frac{31}{32}$ in. pitch circle diameter to depth so that the bottom of the holes coincide with the top edges of the transfer ports in the cylinder liner. The bottoms of these holes were rounded off by grinding the drill hemispherical to ensure smoother gas flow through the passages. It will be noted that the tenth hole which comes

stresses, without causing ovality on release, when boring the cylinders by mounting in a vee block on the boring table. Consequently, the liners were machined at one setting by cutting the cast-iron rod of sufficient length to use one end as a chucking-piece. The outside diameter was made of such a size so that they were a fairly tight fit in the bores in the port belt previously made to receive them. This was done by trial and error while the portion beyond the flanges, where this method could not be employed,

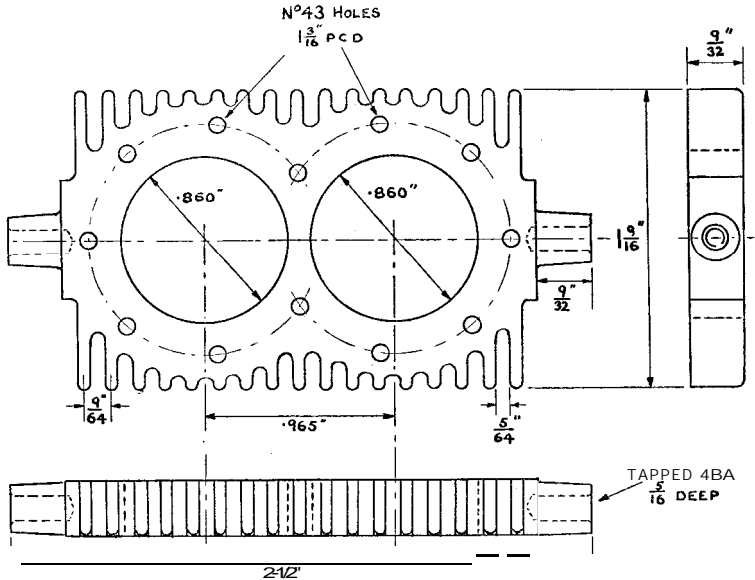


Fig. 18. Cylinder cooling fins

directly between the two cylinders has been omitted to ensure that the centre distance between them is as small as possible. A suitable block of duralumin was similarly prepared for the cylinder cooling fins and is shown in Figs. 17 and 18. This, together with the port belt, was secured to the faceplate and the two bores to take the cylinder liners were machined to a diameter of 0.860 in. It is essential that the centres are exactly the same in the port belt as in the block for the cylinder fins. The boring of the hole to take the transfer cylinder liner cuts the holes previously drilled for the transfer passages so that one side of these passages is formed by the outside wall of the cylinder liner. The object of the lands between the transfer passages is so that the bottom end of the liner shall be adequately supported. The bore for the exhaust cylinder was machined after sliding the sandwich along the faceplate by the required amount. No facing of the ends was attempted, since the bores are automatically at right-angles to their ends.

The cylinder liners are shown in Figs. 19 and 20 and were machined from 1-1/8 in. diameter 1 per cent. nickel cast-iron bar. Since the bore is 0.740 in. diameter, the wall was not considered sufficient to withstand the clamping

was made exactly the same size. It will be remembered that the bores in the port belt and finned block were machined at one setting while clamped together. Fine circumferential lines were scribed in the lathe to mark the upper and lower limits of the ports and dividing into ten equal parts by fine longitudinal lines. After parting off to the correct lengths the liners were mounted, separately, on a mandrel and clamped securely endwise up against a shoulder. One side of the collar of one liner was then machined, and without moving the longitudinal position of the tool, the similar face of the collar of the other liner was faced up. This method was repeated for facing the other sides of the two collars. This ensured that the thickness of these is exactly the same, although the actual dimension is not of great importance. The bores of the liners were then lapped, using non-expanding aluminium laps, two of which were required; one for roughing and the other for finishing. Moles were next drilled for the ports using a drill, the diameter of which is slightly less than the depth of the ports. This was carried out by gripping the liners endwise in a vice on the vertical slide, centring with a centre drill and using the three-jaw chuck to hold the drill. This method, it

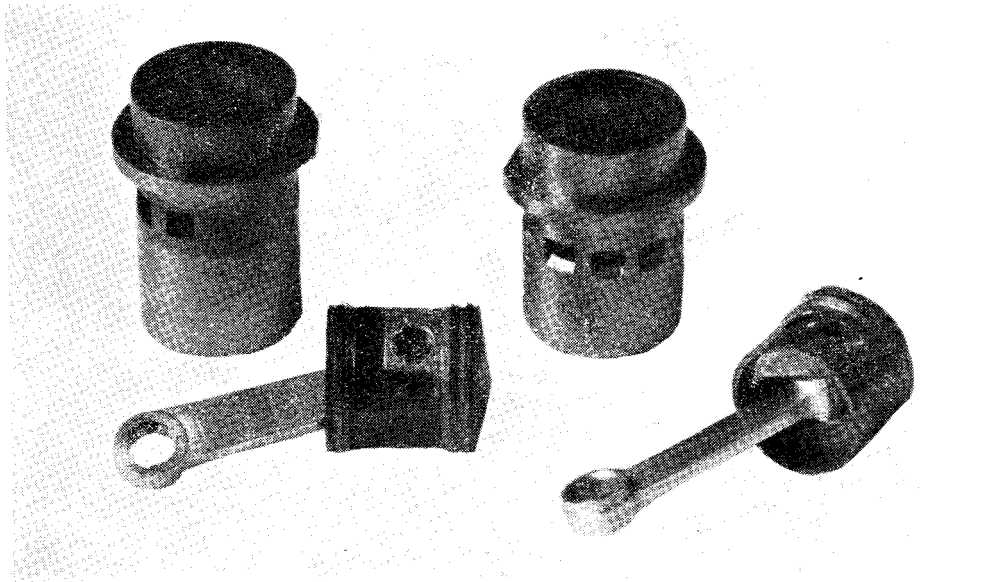


Fig. 19. Cylinder liners, pistons and connecting-rods

has been found, gives more accurate location to the holes and, if a fine feed is used, lessens the amount of burr caused by the drill on the break through. It also obviates the need for centre-pops which are impracticable on thin cast-iron shells. The circular holes so formed were squared out by hand, using a small square

file leaving a land of about 1/16 in. between each port. The bottom external edges of the transfer ports were bevelled off at about 45 deg. to ensure smoother gas flow. The cylinders were finished off by removing the slight burr, raised from the filing, with a piece of fine emery-paper on the finger.
(To be continued)

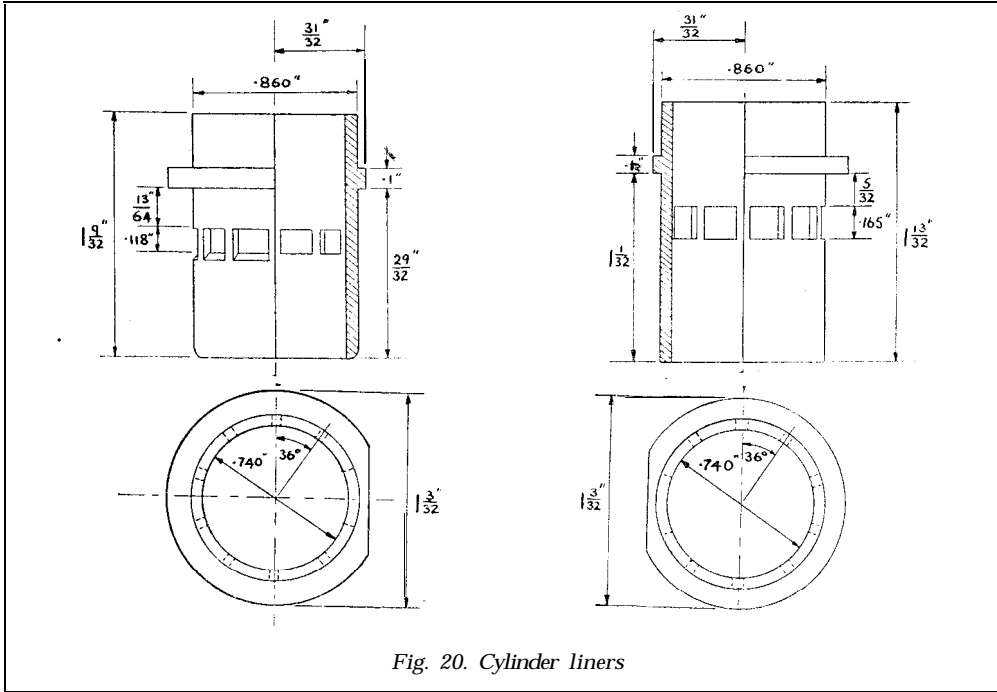


Fig. 20. Cylinder liners

*A Split-Single Two-Stroke Engine

An efficient unit for propelling a class "C" Hydroplane

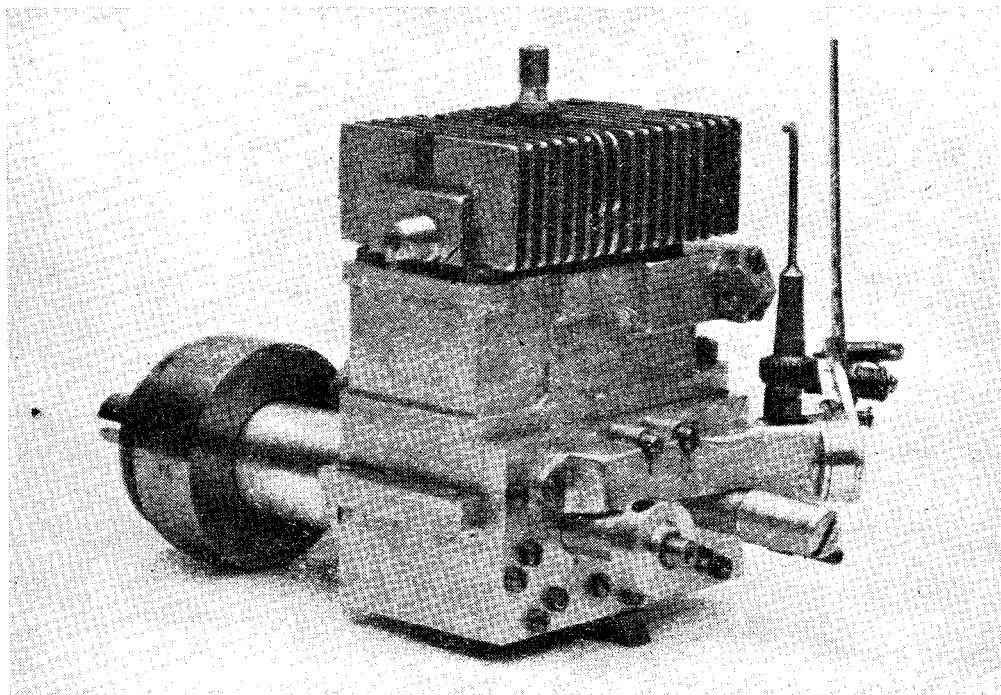
by R. E. Mitchell

THE pistons were next on the list and are shown in Figs. 19 and 21. These were machined from cast magnesium alloy bar. This material has been used in all previous engines and has the advantage over the aluminium alloys in that its density is considerably less.

This enables the reciprocating weight to be kept to a minimum, but against this is the poor corrosion resistance of magnesium alloys but it appears to be adequate. The transfer piston is comparatively short to ensure a good gas flow from the crankcase to the transfer ports. In this design it does not matter if the transfer ports are uncovered at top dead centre. The inside of the piston is machined away 2s much as possible to reduce weight. The exhaust piston is made slightly longer than the stroke in order to accommodate the crankcase com-

pression ring at the bottom of its skirt. Since the M.P.B.A. rules demand a silencer, no attempt was made to increase the volumetric efficiency by allowing the piston to uncover the exhaust port at top dead centre. The exterior of the exhaust piston is not in communication with the crankcase, so the weight reduction in this case was carried out on the exterior of the piston. Leaving the interior of the piston solid except for the connecting-rod slot helps to increase the crankcase compression ratio. The clearance between the piston and the cylinder is rather more than usual being of the order of 0.002 in. Since the ports are operated by piston crowns, compression rings, of which there are two per piston, are placed as near 2s possible to the tops of the pistons. The piston skirts, since they play no part in isolating the transfer ports from the exhaust ports, are considerably relieved except for lands at the top and bottom. The piston ring grooves are machined by using

**Continued from page 551, "M.E.," April 24, 1952.*



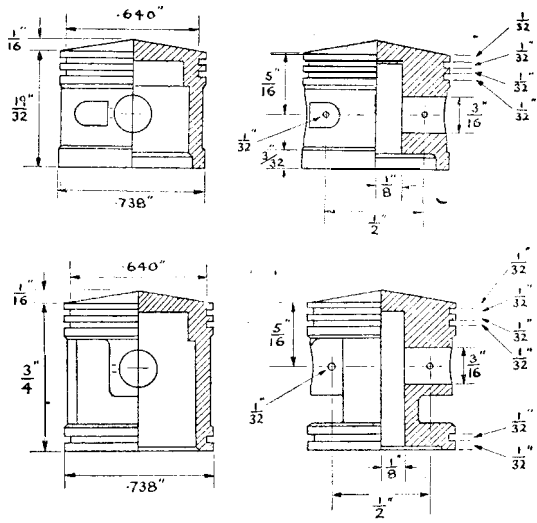


Fig. 21. Pistons

a parting tool, leaving a groove whose width is 0.033 in. This tool is used for piston ring grooves only. The obtuse angles for the conical crowns were left when parting-off the pistons. The holes for the gudgeon-pins were drilled and reamed 3/16 in. diameter by mounting the piston on an angle-plate on the vertical slide. It is very important that the gudgeon-pins are at right-angles to the axis of the piston, and careful setting up is required so that this condition is realised. The piston rings, which are of about 1/32 in. square section, are of 1 per cent. nickel cast-iron as used for the cylinder liners. They were prepared by the usual method of roughing out a blank which is lapped to the correct

thickness after which it is slotted and then compressed on to a mandrel with the gap closed before turning the outside diameter to the correct size. Fig. 22 shows the gudgeon-pins which are a straight forward turning job from 3 per cent nickel-steel bar and are 3/16 in. diameter with a 1/8 in. diameter reamed hole through the centre. No end pads are used. In a previous design serious scoring of the cylinder bore had resulted from their use. To provide endwise location, the piston and pin are drilled 1/32 in. diameter at each end to take a piano wire pin. The gudgeon-pins were finally case-hardened.

From the components so far finished the connecting-rod centres were checked. It will be noticed that the big-end eyes are not symmetrical. This is to provide clearance for the balanceweights on the crank discs. Also, the web is not symmetrical to allow clearance between the rod and the crankcase which is more on one side than the other due to the employment of desaxe cylinders. The connecting-rods, as shown in Figs. 19 and 22, are made from duralumin.

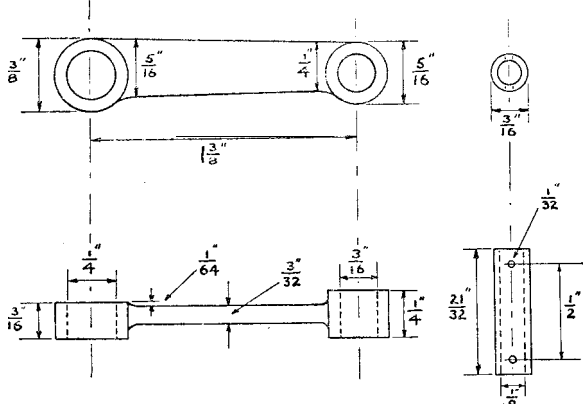


Fig. 22. Connecting-rods and gudgeon-pins

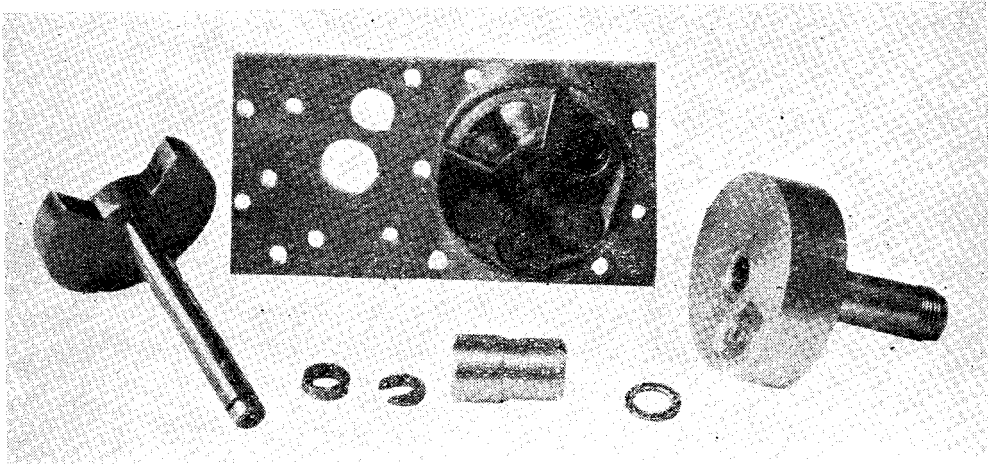


Fig. 23. Rotary inlet valves

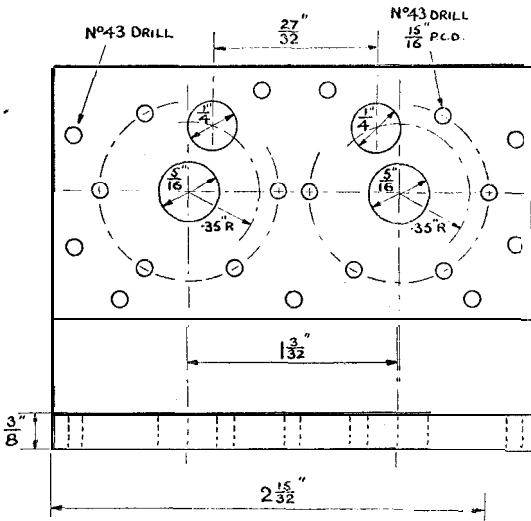


Fig. 24. Crankcase backplate for rotary inlet valves

The holes for the big and little-ends were first bored and reamed, taking great care to ensure that they would be parallel to one another. Excess metal was removed by milling and filing. The usual "I" section is not given to the web owing to the need for keeping this as small as possible. The connecting-rods, it will be noticed, are mirror images of each other.

At this stage, the components so far made were assembled and the dimensions were checked for the rotary valve assemblies which are shown in Figs. 23, 24 and 25. The end plate consists of a rectangular plate of dural 3/16 in. thick and machined on all faces; the external dimensions, at this stage, being slightly larger than those of the crankcase. Two 5/16 in. diameter reamed holes were made in the required positions and at the same centre distance as the crankshafts. These were made by mounting the work on the vertical slide and using the cross-slide index to ensure the correct centre distance. The stationary parts of the valves is a straightforward turning operation with the outside diameters being finished to size by mounting on a 3/16 in. diameter mandrel. It is very impor-

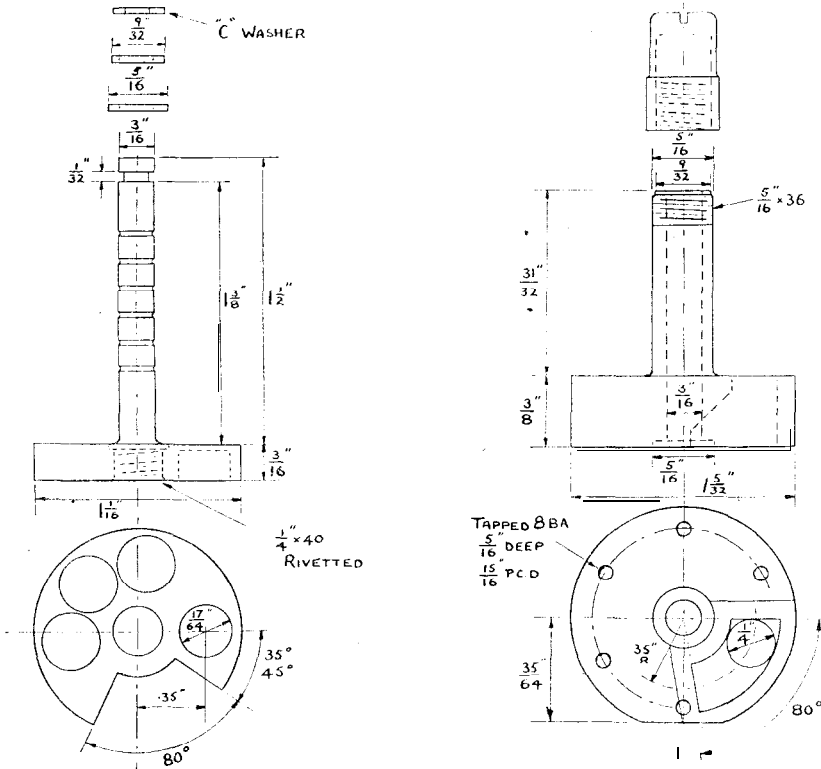


Fig. 25. Rotary inlet valves

tant that the valve face is exactly at right-angles to the bore, and absolute accuracy cannot be guaranteed by drilling a 3/16in. diameter hole of this length. The major diameters were made about 0.0005 in. smaller than the crankcase bores previously made to receive the rotary valves. Since these bores overlap, a flat had to be milled on each to enable them to be fitted.

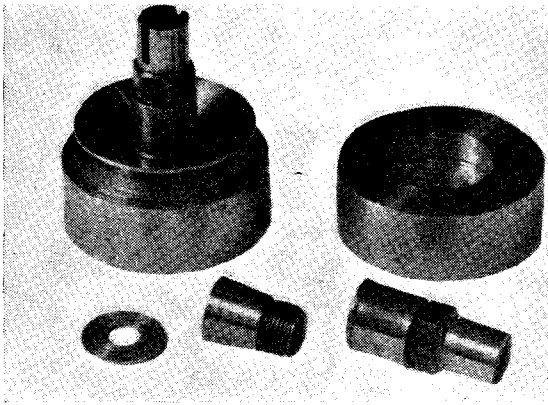


Fig. 26. Flywheels and collets

was removed and the diameter was reduced by 0.0005 in. An oil hole was also provided for extra lubrication. One of the spindles was allowed to extend for 3/8 in. beyond its bearing with the idea of providing for a magneto drive. The other is located endwise by a "C" washer in a circumferential groove and retained by a screwed cap.

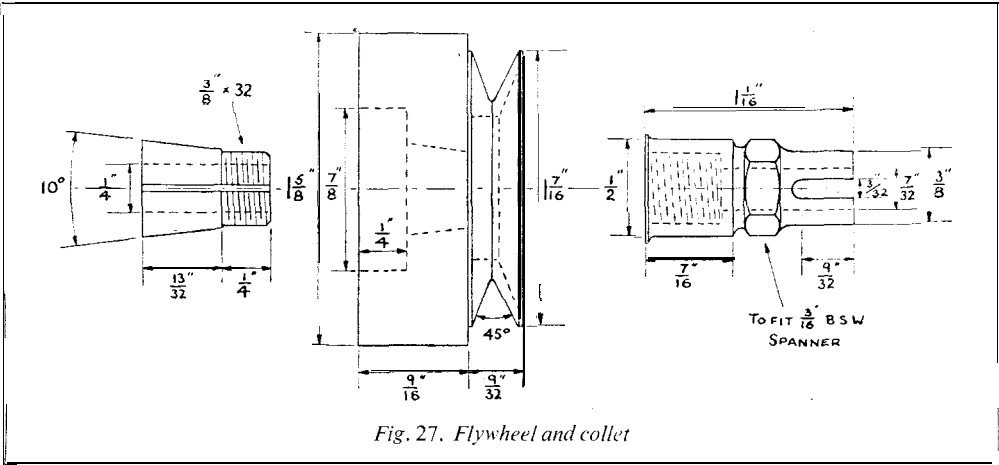


Fig. 27. Flywheel and collet

If a slight error has been made in the centres of the two 5/16 in. diameter holes in the end-plate, a little scraping can correct this because a gastight joint does not have to be made here. The rotating parts of the valves have mild-steel discs on 3 per cent. nickel steel spindles. These were connected together by screwing and riveting over before finally turning to size. The whole of these parts would have been made from nickel-steel had a piece of suitable size been available. It was considered that the extra strength of nickel-steel over mild-steel would be needed on a spindle of only 3/16 in. diameter. It must be emphasised that suitable radii should be left at all changes of section and given as good a finish as possible to reduce the risk of fatigue failure. It may be mentioned here that originally the spindles were made what was considered to be a good running fit in the bearings, although possibly on the tight side with the hope that they would run themselves in. On first running the engine, however, one of the spindles seised, although not seriously. The pick-up on the spindle

The cut-away portions of the valves were removed by saw and file. It was intended that balancing should be effected by drilling three blind holes nearly through the thickness of the disc and each was counterbored slightly to receive a disc of brass 0.010 in. thick which was soft-soldered in position, so as not to reduce crankcase compression. This, however, did not achieve complete balance, so as much metal as 'possible' was filed away from this region, and even now they are not completely balanced. The flywheels, which are shown in Figs. 26 and 27, were turned from medium carbon-steel salvaged from an old drilling machine spindle. The diameter of the wheels is governed by the centre distance of the cranks, and doubt was expressed as to whether they would be heavy enough at 4-1/2 oz. each and a diameter of 1-5/8 in. Subsequent running has proved that the fears were groundless. Most of the weight is concentrated in the rim and the back side is counterbored to place as much weight as possible over the bearing and to prevent overhang. The

bores are made taper with a total included angle of 10 deg. The groove for the starting cord is machined integral with one flywheel. Most starting pulleys appear to be made too large in diameter to give the engine the required spin to start it, but if the groove is made with an angle of not more than 45 deg. no slipping of the belt is experienced. The collets are machined from 3 per cent. nickel-steel left in normalised condition and have a 10 deg. taper to correspond with the flywheel bores. They each have one longitudinal split. The clamping nuts, one of which, of course, has a left-hand thread, also form the female portions of the inboard universal joints and are case-hardened, but the threaded ends are tempered back slightly. This was carried out by standing the nut in water, covering the portion to be left hard and playing a flame on the part above the water line. The threads are 3/8 in. diameter x 32 t.p.i. and were cut in the lathe. When first running the engine with fixed advanced ignition timing, the flywheels occasionally turned on the crankshafts due mainly to the violent acceleration made possible with the rather light flywheels. Owing to the same cause, similar difficulty was also experienced with the magneto rotor and couplings. Since using a glow plug, however, this trouble has never been in evidence because it is possible to start the engine on a much more nearly closed strangler setting.

Figs. 17 and 28 show the cylinder-head which was cut from solid dural and is machined all

over. Clearance was machined for the conical piston crowns at the required centre distance. Exact measurement here is unnecessary. To connect the two cylinder bores a passage was cut by hand and is approximately 1/2 in. x 1/8 in. cross-section midway between the two cylinders. The head was reversed and drilled and tapped 1/4 in. x 32 t.p.i. to take 5/32 in. reach plugs. The plug joint seating was machined at the same setting. It will be noted that the plug is offset by 1/8 in. towards the transfer cylinder. This was done because the mixture in this vicinity will in all probability be less diluted with exhaust gas, making for easier starting and higher flame speeds through the mixture. The last operation on the finned belt was to machine a lug on each end which is tapped 4 B.A. These are to take tie-rods to enable the engine to be more rigidly held in the hull and to distribute some of the load imposed on the crankcase holding-down lugs, the bolt centres being only about 1 in.

The induction ports in the end cover of the crankcase are each 1/4 in. diameter, which may be considered small by modern standards. There is, however, sufficient metal for these to be opened up although this has not yet been attempted. It was rather a problem to design a suitable induction manifold. Two carburettors were considered but were abandoned because one would foul the magneto if it was ultimately decided to use this form of ignition.

(To be continued)

For the Bookshelf

The Beauty of Old Trains, by C. Hamilton Ellis. (London: George Allen & Unwin Ltd.) 148 pages, size 6 in. by 9 in. Eight coloured plates, 60 halftone reproductions. Price 20s. net.

This is a most unusual book, but one that can be read and enjoyed by anyone who knows the fascination of a railway train. To most of the older generation, however, railway trains have now lost much of the beauty that was once theirs, and Mr. Hamilton Ellis has done well to recall and to give something like permanent record to facts which were in danger of becoming lost. He does it most successfully in this, his latest contribution to railway literature, by recounting his early impressions and the modifications which he later thought necessary to make in them.

His well-known style of writing is much in evidence; though he wisely leaves his readers entirely free to decide whether or not they agree with his opinions. But he is always entertaining and frequently amusing.

The eight coloured plates are worthy additions to the still-too-rare visual evidence of the colours of old trains and locomotives, and most of the subjects are uncommon. The photographs, from which the sixty halftone illustrations (which

are printed on twelve inserted art-paper pages) have been prepared, are just right for such a book and were obviously selected with the greatest care. We note two slips in the captions: first, the engine seen in Fig. 35 cannot be the *Albert Edward*; the trailing hornblocks show that she was originally a 2-2-2 later altered to 4-2-2. Although it is difficult to decide precisely from scrutinising a halftone reproduction, the number appears to us to be 3021, which would mean that the engine is actually the old *Wigmore Castle*, one of this class which was originally built on the 2-2-2 wheel arrangement. The second slip concerns Fig. 54; the 4-4-2 tank engine seen in this picture must be No. 1126; the G.C.R. locomotive bearing the number 126 was a 0-6-0 tender engine designed by Parker and built about

We are interested in Mr. Ellis's comments concerning the marked similarity between the Great Northern locomotives and stock, and those of the Great Northern of Ireland. The fact is that J. C. Park, who introduced English Great Northern features to the G.N.R.(I) was formerly a draughtsman under Patrick Stirling at Doncaster; Mr. Ellis was evidently unaware of this.

All the same, this delightful book seems destined to find a place among the classics of its kind.

*A Split-Single Two-Stroke Engine

An efficient unit for propelling a class " C " Hydroplane

by R. E. Mitchell

ANOTHER objection to twin carburetors is due to their being displaced laterally by $\frac{27}{32}$ in. Due to centrifugal force, fuel feed to the inner carburettor may cease altogether, and would not probably be compensated for completely by the other, thus upsetting the mixture strength after a certain speed had been reached. A manifold in the form of a "Y," using a single carburettor, was considered but had to be abandoned, again because of fouling the magneto. If this

is $\frac{1}{32}$ in. diameter with two similar holes communicating with the choke. This type of jet is not usually considered to be the best from the performance point of view. The more usual type whereby only the tapered portion of the needle obstructs the gas flow has since been tried, but an improvement in performance was not detected. One disadvantage, however, was noted in that the needle setting for optimum results was made far more critical ; too critical

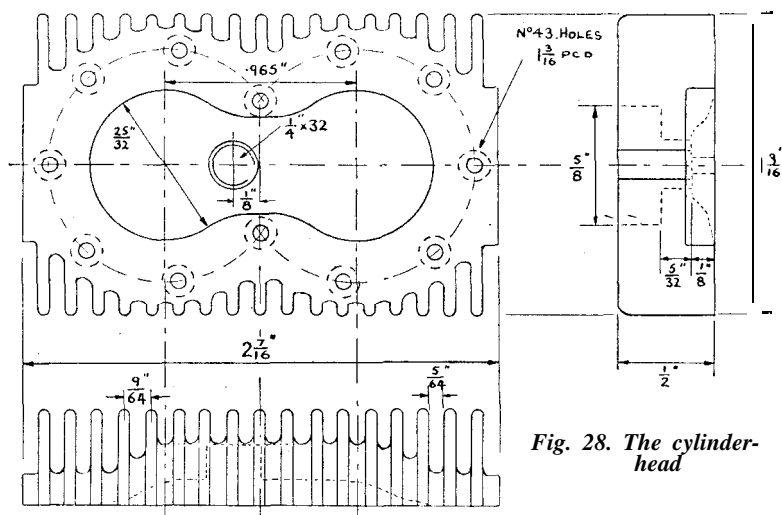


Fig. 28. The cylinder-head

could have been avoided the carburettor too near the magneto would probably result in the latter being sprayed with fuel. Although not ideal from the gas flow aspect, the final design decided on is as shown in Figs. 29 and 30. The carburettor is cut from the solid and is of dural. The choke is of oval cross-section and measures $\frac{19}{64}$ in. x $\frac{1}{4}$ in., the major dimension being in the horizontal plane. Since with a two-stroke engine having rotary disc inlet valves, the carburettor is very low in the hull, making it rather inaccessible, it was decided to use a jet which can be removed by unscrewing a nut on top of the venturi. The jet assembly is best shown by an illustration and is shown in Figs 29 and 30. It is made of brass and the jet proper

to ensure repeatable performance and was abandoned. The needle is of $\frac{1}{16}$ in. diameter silver-steel soft-soldered into a brass nut which is milled on the outside to engage with a bronze leaf spring anchored to the carburettor body with two 10-B.A. screws. Some designs rely on the friction of tight screw threads to hold the jet needle in position, but this is considered to be a rather crude method, being very susceptible to movement due to engine vibration.

Most engines of 10 c.c. capacity and less are not fitted with any means of controlling their speed. In aircraft practice the propeller does, however, provide a suitable load. One of the surest ways of wrecking an engine is to allow it to attain the high speeds of which it is capable without a load. Under these conditions the centrifugal loads applied, particularly to the connecting-rod, are far in excess of the normal

*Continued from page 570, "M.E.", May 1, 1952.

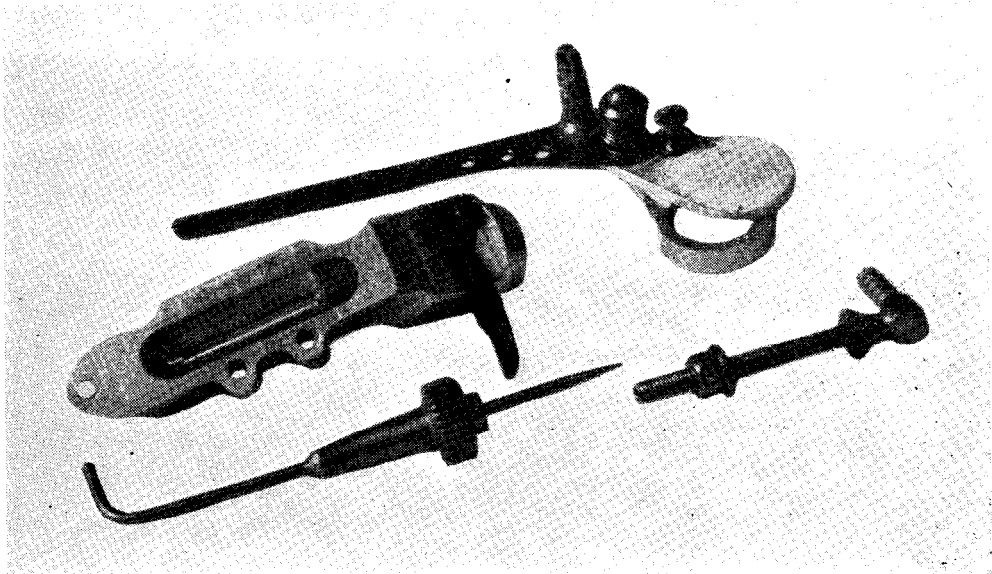
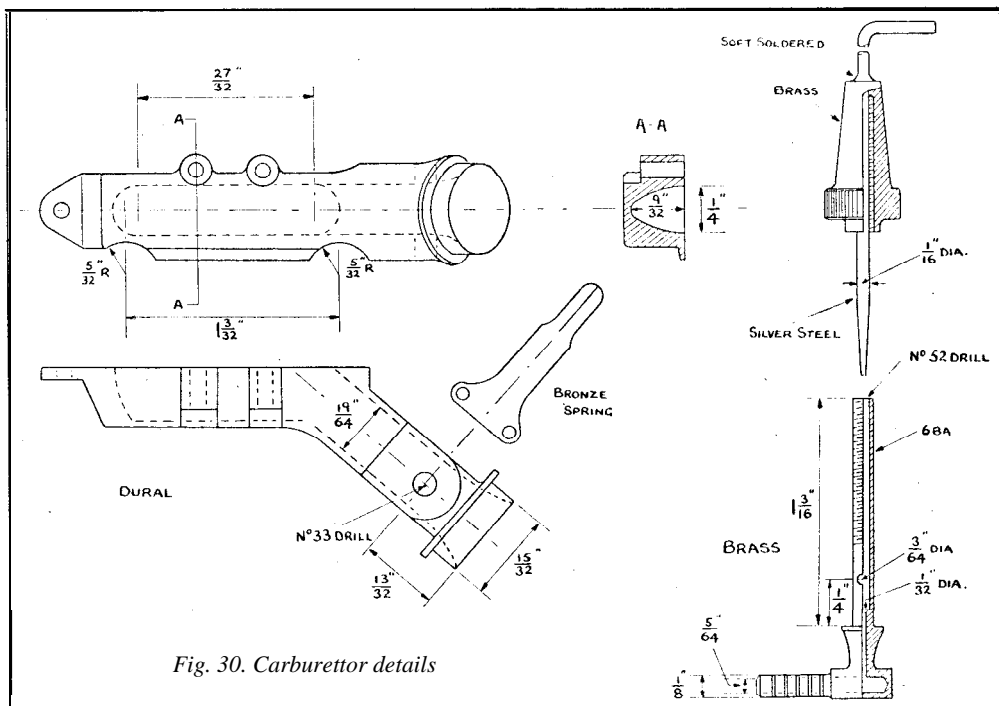


Fig. 29. The carburettor

working stresses. This can easily be avoided by the fitting of a simple strangler consisting of a disc pivoted so that it can be moved across the carburettor inlet, thus obstructing the gas flow. This method has been used successfully on four-stroke engines using spark ignition, and works

equally well in the present two-stroke case.

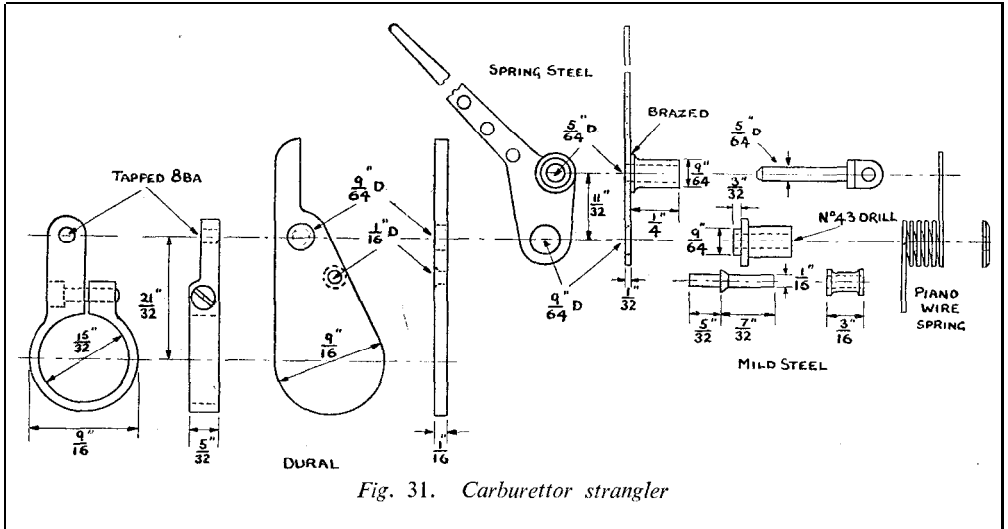
The usual idea that a glow-plug engine is capable of little or no speed control has been found to be erroneous. Poor carburettor design is likely to be mostly responsible. Since the boat in which this engine is installed is fitted with



a device operated by water pressure in the case of submergence, the strangler disc and its lever are independent although pivoted on the same spindle, as shown in Figs. 29 and 31. Connection between the two is made by means of a peg which is withdrawn when the knock-off device is operated, so allowing the spring-loaded disc to cover the carburettor intake and prevent-

into the dural do strip, most of the holes being at least three diameters deep, there is sufficient space for them to be tapped out to a larger size. Dismantling an engine except for repairs or alterations should not be carried out, because the alignment is bound to be adversely affected on reassembly.

All the joints, which are fairly wide, are metal



ing a charge of water entering the crankcase. Previous experience with water entering a high revving engine has shown that the results can be serious. This is also used as the normal stopping method, and although instruction leaflets accompanying several commercially made engines state that this method must not be used, no harm appears to have arisen from its use. As a matter of interest, the same method has been used for stopping a four-stroke engine with a compression ratio of 14-1/2 : 1 where the chances of a hydraulic lock are much greater.

The drilling and tapping of the holes required for holding the various components together was the last operation. The housings for the main bearings are each secured to the crankcase by five screws, the rear crankcase cover by nine screws and the stationary parts of the rotary valves by five screws each. All these screws are 8 B.A. and have cheese-heads. The port belt is secured to the crankcase by four through-going 6-B.A. hexagon-head screws and two 6-B. studs. The studs are used beneath the exhaust ports. The cylinder-head is held down by twelve 8-B.A. cheese-head screws which pass through the finned belt into the port belt, the holes being deeply counterbored to receive the heads. The use of screws in preference to studs and nuts may be regarded as taking the easy way out and is the first design in which they have been used so extensively. Studs have been used in a four-stroke design from which the nuts have never been removed in the four years of its existence. If, however, the threads tapped

to metal, no gaskets or jointing materials whatever are used.

When first attempting to start the engine a magneto driven from one of the rotary valve spindles was used. The engine stopped after a few revolutions and the trouble was found to be due to the magneto coupling having slipped on the shaft. The flywheels are rather on the light side and the initial acceleration was probably the cause of the trouble. Attempts to remedy the defect failed, and since the power boat season was approaching, the spark ignition was abandoned in favour of a glow-plug. Fig. 17 shows the glow-plug connector which is an improvement on the usual crocodile clip. It consists of a piece of 1/4 in. thick ebonite sheet provided with two bronze leaf springs which grip the cylinder-head. It is located laterally by the ends of the screws securing these springs engaging in the cylinder-head cooling fins. The switch also contains two bronze spring contacts which normally close on the live terminal of the plug. To disconnect the plug from the battery, the contacts are opened by a cam placed between them.

A 2 V Exide "Gel-Cel" accumulator is used and the length of the leads is adjusted so that the voltage at the plug when on load is 1.5 V. With this arrangement, no glow-plug has been burnt out during the whole of the season.

After running the engine for a few minutes one of the rotary valve spindles seized. This was easily removed by warming the dural bush. The "picked up" metal on the spindle was removed and the valve was replaced. The

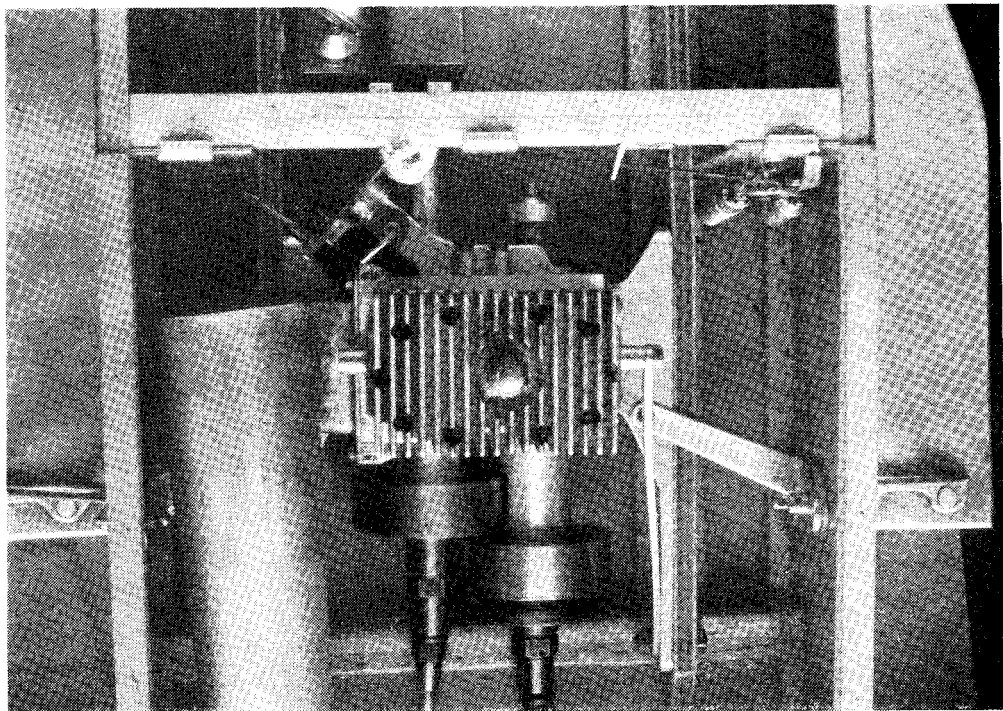


Fig. 32. Engine installed in the hydroplane " Gamma "

engine appeared to behave itself, but the speed of the hydroplane showed signs of dropping.

This was at first thought to be due to piston ring flutter on account of the light ring pressures which were being used. Replacing these with rings having a higher pressure resulted in no detectable improvement. However, when trying the engine with new piston rings, the same rotary valve spindle again seized. Both spindles were removed and a further 0.0005 in. was taken off their diameters. The speed of the boat was thereby increased and no further trouble was experienced on this score.

No attempt has been made to determine the power output, since a dynamometer to absorb the power from the two shafts would be difficult to devise. To take the power from one shaft only would mean that the whole of the output from one cylinder would be transmitting through the synchronising gears, a thing which it is desirous to avoid. Also, no attempt has been made to measure the engine speed when running light, since this only imposes undue stresses on the engine to no purpose.

The final photograph, Fig. 32, shows a plan of the engine as installed in the hydroplane Gamma.

A High-Pressure Table-Engine

(Continued from page 595)

and links, the governor controls a butterfly-type throttle valve mounted immediately in front of the steam-chest.

Finish

All bright parts of the model are polished to a high degree, using various grades of emery down to "blue-back," and plenty of that useful commodity, elbow-grease. Other parts are a pleasing shade of green, with a stone-coloured foundation and stained and polished base-board.

Normally a glass case is used to cover the model, but it may be worked under compressed air.

Speaking personally, my chief regret is that I have not had the pleasure of seeing it working. It must be a very pleasant sight, at that stately gait of one revolution in two seconds, and to me, at least, would be worth twenty times the sight of a buzz-box batting round an arena at ninety miles an hour! It's a good job we aren't all alike, isn't it?